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SHOCK AND VIBRATION TESTING - CURRENT THEORY AND UNCERTAINTIES

By

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SHOCK AND VIBRATION TESTING - CURRENT THEORY AND UNCERTANTIES

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ABSTRACT

The report presents a review of current shock and vibration testing technology, based upon literature study, conferences with testing personnel and shock and vibration experts, and the personal experiences of the author. The technological frontiers of the many aspects of shock and vibration testing are outlined. Those areas of research which would most significantly improve reliability or economy are indicated. The report also indicates those practices and theories which are not based upon fact and necessarily involve uncertainties.

INTRODUCTION

This is a report on NCEL Work Unit Number 63-006: Development of Improved Dynamic Test for Structures, being performed for the Naval Electronics Laboratory Center. The main purpose of this report is to describe the initial findings of the literature survey and to summarize the information gained in interviews with experts in the field of shock and vibration testing. The report reflects the depth and breadth of the study to date and separates out the various problem areas, indicating the state-of-the-art in each area. It should provide a convenient point of departure for discussion with cognizant NELC personnel of future orientation of the study and the points which must be emphasized.

The report is organized as a broad review of the topics pertinent to dynamic testing especially in the light of the apparent requirements for shipboard equipment. An effort was made to place no particular emphasis on any one topic, but rather to establish the state-of-the-art and understanding in each. Few, if any, conclusions are reached or were sought at this time.

Dynamic Tests

Dynamic tests are given to equipment to assure that it can operate in and survive its expected dynamic environment. The test itself is most vital for it is made a mandatory acceptance condition. As such it motivates equipment manufacturers to build in dynamic sufficiency commensurate with the test. An unduly severe test unnecessarily increases equipment cost and often weight, while too weak a test requirement will not provide sufficient dynamic reliability. For the shipboard situation the environment is quite well understood, probably better than most of the missile and space environments, but still far from perfectly.

Dynamic environment can be organized into several classes, the main ones being boradly described as shock and vibration. Two newer shock-like environments must now be considered; a low level repetitive shock such as may be cuased by rapid firing cannon and an air blast shock due to the blast wave emanating from large cannon and near miss atmospheric explosions. The unifying characteristics of loading in various dynamic environments is the fact that the mass or inertia of an item subjected to dynamic loading plays a significant role; the inherent mass-spring quality of matter allows inertial overshoot or ringing so that the actual loading is not a simple constant factor product of the applied load.

There appears to be a slight semantic problem in technically differentiating between transient vibration and shock. Actually transient vibration is a more general term which includes shock. The word "vibration" as used in this report refers to dynamic equipment oscillations that are of a more or less long term steady state nature.

The problem of specifying and performing the most efficient dynamic test is very difficult. In general the governmental and industrial practitioners admit and bemoan this difficulty and the current only partial understanding of how to cope with it. The field, however, is extremely complex and needs further study, far in excess of what the government or industry is willing to support at this time. Consequently, in the immediate future, one shall have to be content with only partial understanding of the dynamic processes involved.

The total problem is humbling. It can be broken into thirty or more subproblems, almost all of which are only weakly solvable under a crippling array of conditions or "if's." The deductions which can be made from "hard" starting points to "hard" results all seem of little importance when compared to the seemingly innocent question of "How should we test this?" On the following pages an attempt is made to organize the findings of the literature search and study which should allow an orderly review of the existing knowledge, preparatory to defining an improved dynamic test.

Literature Survey

The available literature is extensive, the largest repository being the "Shock and Vibration Bulletin" published by the Naval Research Laboratory (NRL). This source is certainly an advantage to students of oscillatory dynamics, however, it does tend to be a little wordy and circuitous. The proceedings of the Institute of Environmental Sciences (one volume published each year) also constitutes a valuable source. Beyond these two, several publications are quite thorough and contain summaries of the state-of-the-art up to the time of their publication; notably the works of Harris and Crede,[1] Barton,[2] Crandall,[3] and Morrow.[4] In fact, Morrow's book[4] seems to be the only volume which technically attacks the problems of dynamic testing.

In every substantial study of a field with appreciable history one must decide the extent to which the literature search is pursued, so as to not go beyond the point at which further background ceases to add materially to the objective. It is felt that at this point the task is 0.8 complete; not so much in the number of articles reviewed, but rather in the necessary effort expended.

The Current Specifications

Quite naturally since this report is motivated to propose improved dynamic tests it must constantly point out those aspects of the current procedures which detract from a perfect test. As such, the task forces a criticism. However, as we prepare to point out the shortcomings, it is well to consider that the current tests do accomplish a great deal, and do vastly improve the dynamic reliability of tested equipment. This task, however, is not to praise the status quo; it is intended to point

out the extreme difficulty of the problem in general and to specifically set forth the multitude of problems that still remain unanswered.

We shall not dwell on the current practices at this time. All who read this document are well acquainted with those specifications. Let us proceed directly then to review the recent history of dynamic testing to note the significant gaps of knowledge.

The Traditional Defense of Current Navy Specifications

The Navy must periodically defend its specifications from those who are forced to comply with them. It is interesting to consider the arguments they have found effective in placating criticism. Most generally these arguments have had to defend the shock specification requiring the High Impact Shock Machines.[5] Fortunately, they have not been forced to defend the vibration specification[6] with its peculiar concept of test at integer frequency values, undoubtedly becuase it is a much less severe requirement.

Reference 7 presents a typical justification of the Navy Specifications, MIL-S-901[5] and MIL-STD-167.[6] The traditional argument is that the specifications have "evolved" over a long period of time and that they duplicate the damage actually observed aboard ship. This may be questioned. There has been virtually no evolvement of the shock specification since it was developed or copied from a British design in the early 1940's. An oblique fixture was added somewhat later to include some transverse loading on the medium weight machine. The author also cites one of the limitations of the shock machine as he comments about potential clever use of absorbers designed specifically to defeat the shock machines. Vigness, in a discussion of the shock machines,[8] explains that the lightweight machine was copied from a British design in 1940, and that the mediumweight machine was designed and built in 1942. In his careful discussion of the history of these machines, one notes no further discussion of improvements that have been made in these machines since that time. In this 1961 report, Vigness states, "The science of shock testing is still in a youthful state and that considerable judgment may be necessary in formulating tests, and that considerable changes may be expected in shock machines and procedures in the future." He then gives the justification that the tests seem to produce the same type of damage actually seen in shipboard shocks. Bort[9] also repeats the explanation on page 467 of his article, but on the next page does a reasonably complete job of covering the machine limitations; for example, he cites the following problems: the direction of the hammer blows are not always realistic, the displacement is abruptly stopped, the velocity is limited, the foundation natural frequency is always set at 60 cps on the mediumweight machine even though the proposed installation may be different, and no allowance is made for the different environments at various locations throughout the ship.

It is now beginning to appear that this situation will change. Several different Navy personnel commented at the recent shock and vibration symposium that changes are on the way; committees have been formed and studies of possible improvements have been initiated. It is hoped that NELC and NCEL can contribute to this progress.

SHOCK

While neither shock nor vibration testing are completely understood, nor are they likely to be in the very near future, many consider shock the more difficult problem. Theoretically, at least, the shock problem is a good deal more difficult and time consuming to accomplish in any given situation, generally because shock implies a transient response while vibration usually implies some periodic response. In actual fact, either problem can be equally difficult depending on the extent of the analysis. Theoretical understanding of the linear problem is well developed, and computer methods can accommodate some very complex nonlinear situations. The Navy through its Structural Mechanics Branch at NRL has for many years supported extensive theoretical study of shock and through this effort has come most of our understanding of the normal mode analysis of the linear problem. Further, the Navy has been the sponsor of the Shock and Vibration Symposia which are the most important meetings where dynamic problems with actual hardware are considered. There has been no apparent effort, however, to establish the credibility of the specified shock tests to the environment either theoretically or experimentally. Practically all equipment shock and vibration testing undertaken in the USA is related to naval, missile, space and aircraft applications. Because of the classified nature of the dynamic loads and the huge expenditure for even the most simple of experiments and the associated data analysis, testing is mostly confined to a few governmental and industrial laboratories with only token participation by the universities and research institutes. There has been some independent experimental effort, by Oleson,[10,11] and Clements,[12] but this has been more directed at proving the accuracy of the theoretical findings rather than pursuing modern effective test procedures. Indeed, most information directly pointed at equipment test appears incidental to the writer's main purpose, i.e., the reporting of some test according to some specification. On completion of a particularly hard task, the investigators write up their difficulties and suggestions for the future, often so specifically, that the material has little application to formulation of new test requirements and goes unnoticed. It certainly appears that this gap should be filled by a team dedicated to continuing effort in this area. Long term, serious, thorough studies should get underway that would, in detail, study dynamic tests and their specification and the ramification of such on the long term reliability of equipment. While testing itself may not initially appear to possess sufficient sophistication to entice the researchers, it will ultimately require deep study.

Invariably, such research will require expensive experimental support. It must, none the less, be undertaken to remedy the present ambiguities.

The Nature of Shock

For the purpose of this report, shock is considered a high intensity, short duration, transient load, usually such that structural response continues for a time after the loading has ceased. An excellent introduction to the shipboard problem is given by Keil[13] which he documents and illustrates with striking photographs of equipment and ship damage as a result of near-miss shock. He also gives expressions for the excitation rise time and decay due to underwater explosions. Figure 1, taken from Keil's report shows the variation of shock spectra with position on the ship and some actual measured ship shock motions.

The actual pressure loading that delivers the shock to a structure is a relatively uncomplicated function of time as shown in Figure 2 from Keil's report.[13] By the time the shock has been reflected and filtered in the many paths to an equipment attachment location, it becomes complicated and can appear as a random burst. Vigness[3] comments on the nature of shock by saying, "Except for time duration there is little difference between a shock motion, as observed under field conditions, and a random vibration. In fact, the random vibrations normal to transportation can be considered as a result of a large number of closely spaced shock excitations. A shock motion, such as is normally observed on structures in the field, is a particular kind of random vibration."

Shock Theory

The type of thinking that will assist in understanding shock test specifications comes from an overview of the theoretical solution of the response of multi-degree-of-freedom systems to transient excitation. Two approaches are used. The first, and the least important for our immediate purposes is the wave propagation approach in which the actual stress waves are carefully followed through the structure. These are generally long and difficult computer solutions, but they have the potential for analysis of very nonlinear response of quite complicated structure including plastic deformation. A good brief discussion of these methods is given by Young in his article in Barton.[2]

The main theory in use by most theoretians working in this area is generally referred to as Normal Mode Theory and is presented in varying degrees of completeness in hundreds of references ranging from Rayleigh's, "Theory of Sound"[14] through Goldstein's, "Classical Mechanics,"[15] to the "Shock and Vibration Handbook,"[1] and virtually every other modern vibration textbook. One of the more lucid expositions is given by Young[2] and several quite complete and detailed developments have been published by the Structural Mechanics Branch of NRL.[16,17,18,19] It is the theory presented in these latter NRL reports thatis the basis for the computations specified in the Dynamic Design Analysis Method (DDAM).[20] The

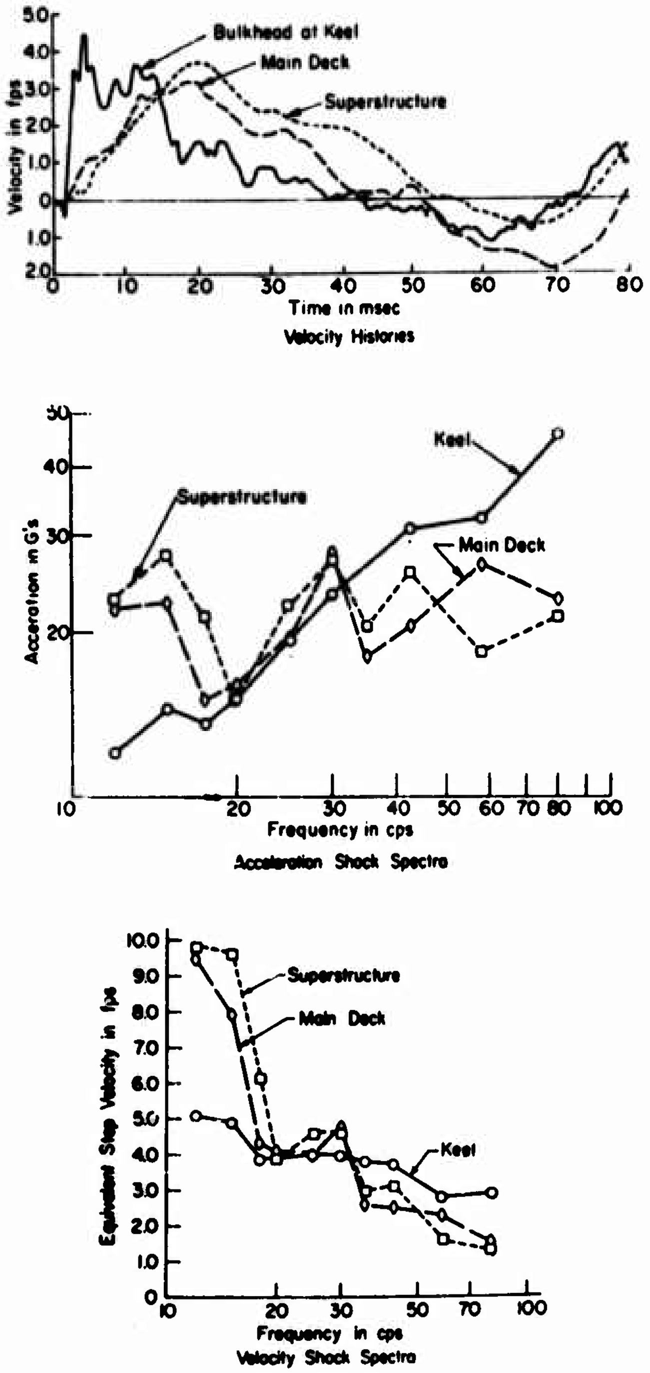

Figure 1. Variation of shock environment throughout a destroyer (for moderate shock)(after Keil[13]).

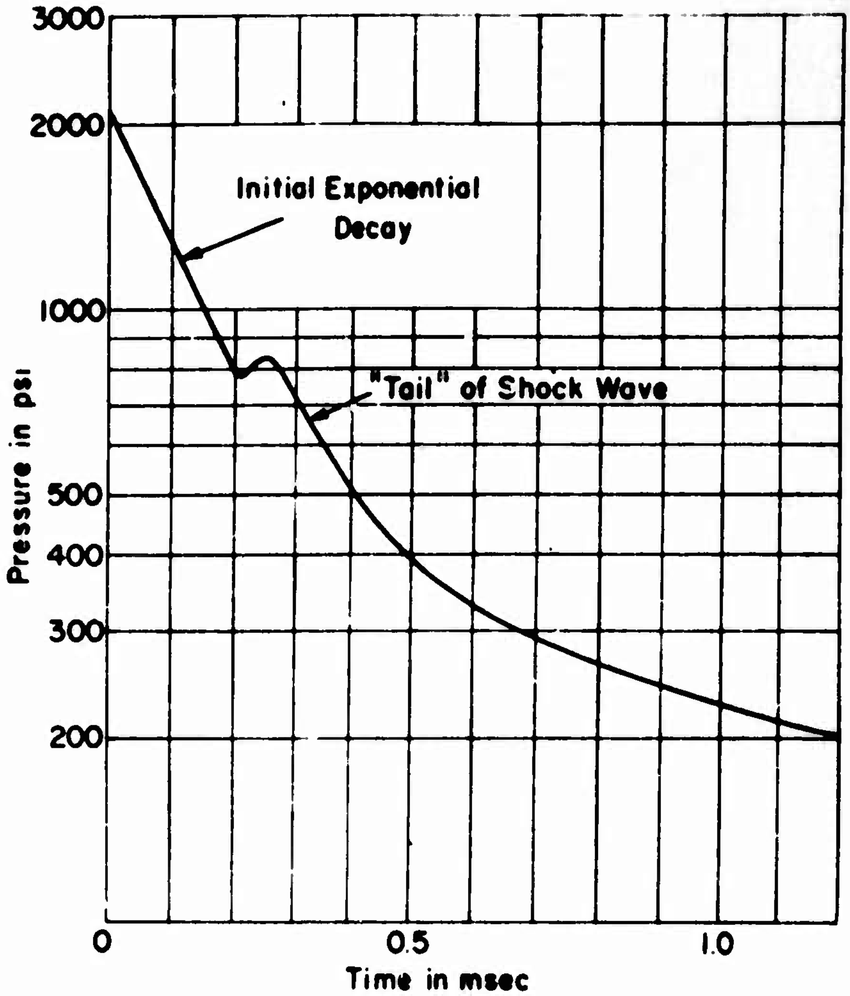

Figure 2. Shock wave from 19-lb TNT at 20 ft (after Keil[13]).

terms "participation factor," "modal mass," and the particular emphasis on transient base excited motions are best developed by the NRL reports. In essence, linear normal mode theory proves that the transient response can be exactly described by a correctly phased summation of the responses of the individual normal modes of the structure. The response of each individual mode is shown to be dependent upon the Fourier Transform of the transient excitation. Such a perfect summation and computation for every normal mode of any real structure (of which there are an infinite number, at least to the extent of individual molecular oscillations) is clearly impossible. Generally speaking current analytical techniques are only able to correctly model the first few modes (sometimes 100) of

any real structure and so the analytician argues that response to modes of higher frequency than he has considered is insignificant. Even so, the correct phasing and summation of response for more than even 5 or 10 modes is seldom accomplished. Rather, it is usually stated that a not too overly convervation estimate of an upper bound to the system motion can be obtained by considering only the magnitudes of the individual modal responses and adding these together in their most unfavorable combination. Much conjecture ensues about this combination and comments on its merits or overly conservativeness are given in a number of references.[2,9,21,22,23,1] Now, if one is willing to neglect the phase relationship and only seeks the magnitudes of the individual modal motions, one need not have the Fourier Transform of the transient input; the simpler shock spectrum will suffice. Hence the popularity of the shock spectrum, and its potential for use by designers.

It may be well to briefly digress on the above mentioned question of the combination of modes. Fung and Barton,[21] in a classic paper on the shock spectrum discuss conditions under which the absolute values of the mode shapes should be added or when the algebraic values may be added. Rubin,[22] also attacks the problem and comes to the conclusion that the errors should usually be less than 10% when utilizing absolute values. Further study of these two papers will increase understanding of the limits of over-estimation of the response under various conditions. The Navy DDAM[20] combines the modes statistically[19] by adding the most highly responding mode to the rms value of several other modes. The wisdom of this procedure has not been theoretically established.

However, on the basis of normal mode theory, if one can deduce the frequency and the mode shape for any particular mode of the structure, one can compute the participation factor and modal mass and then from the shock spectrum determine the maximum amplitude for that mode. The problem as previously mentioned, is combining the response of several modes to determine the total response. Nevertheless, a designer, if he is made knowledgeable of the normal modes and frequencies, could then from the shock spectrum somewhat predict the structural response. As a practical matter the method has not worked out because few people are available who are able to deduce even the mode shapes and frequencies. The DDAM,[20] which attempts to itemize this procedure, has met with difficulty, to the point where still today, many manufacturers would rather submit their hardware to test, since it often costs less.

The Shock Spectrum

As a point of departure for discussion of the shock spectrum, Cunniff[24] gives a definition typical of the many as, ". . . a shock spectrum is taken to be the maximum absolute values of the responses of a hypothetical series of damped or undamped single degree-of-freedom oscillators subject to the shock motion, plotted as a function of the oscillator natural frequencies." Generally if the damping is not stated on the graph, it is assumed to be an undamped spectrum. The idea of the

shock spectrum appears to be credited to Biot.[25] Its early history centered around the famous "Reed Gage" a gage that actually contained a series of reeds of different frequencies together with a method of recording the maximum amplitude of each reed. The device is still in common use for earthquake and high explosives test data recording. Fung's chapter in Barton[2] discusses several other spectra which are extensions of the idea. Morrow[4] in an extended discussion on the shock problem presents the shock spectrum and subdivides it into an initial spectrum (which contains the initial motions of the reeds while the pulse is acting), the residual spectrum (that only contains the reed responses that remain after the pulse terminates) and a positive and negative spectrum which separate the directions of the maxima with respect to the directions of pulse action. These refinements have narrower applicability than the original definition and although they all are useful, the organization of the finer points will not be undertaken at this time. In virtually all of the numerous articles which refer to the shock spectrum, each author gives his own stylized definition of it, sufficient to his needs. For example, all five authors in Barton[2] do so. The concept is just now beginning to creep into the textbooks and shortly a precise uniformly adopted technical definition for it will undoubtedly be agreed upon.

Since the spectrum refers to amplitudes of harmonic oscillations, the product of the displacement and the circular frequency is equal to the velocity, and the product of the velocity and the circular frequency is equal to the acceleration. This simple relationship makes for convenient plotting of displacement, velocity, and acceleration amplitude versus frequency on what has come to be called "four coordinate paper." Figure 3, taken from Vigness' article[8] is an example of spectra plotted in this way.

It must be emphasized at this point that the shock spectrum has a more mathematical background than one might guess from what has been said thus far. It is related to the Fourier spectrum or Fourier Transform of the transient, but contains only about half of the information. Morrow[4] presents a demonstration that, ". . .the residual shock spectrum is simply $2\pi f$ times the magnitude of the Fourier Spectrum . . ." (In a future report this equivalence will be discussed in detail.) The Fourier Transform of a time function is an alternate way of specifying the time function (or transient). The transform yields a continuous plot of frequency information on the pulse, a recipe of how one might construct the transient by an infinite sum of sinusoids, continuously distributed in frequency, each with its own associated phase angle. It is difficult to think of the idea of "continuously distributed in frequency." One may speak of an "amplitude intensity" as a function of frequency; e.g., in per sec per cps. Then one supposedly makes it clear by adding that "the harmonic content" can be comprehended by thinking of the integral of this spectrum between frequency a and frequency b. At any rate, the transform is a complex function of frequency; each frequency yields two values: amplitude and phase, sines and cosines, or real and imaginary

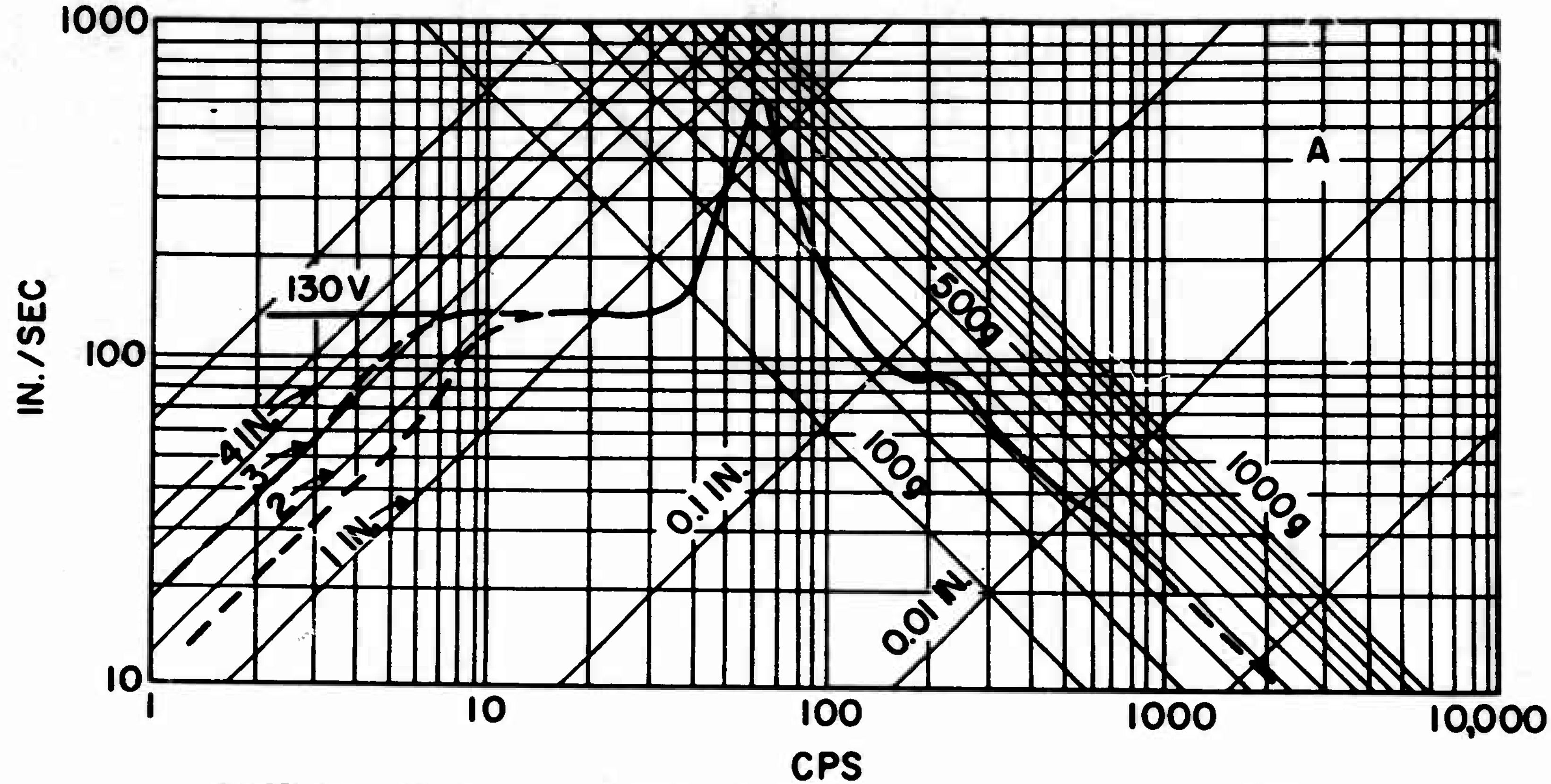

Figure 3. Shock spectrum of motion of a rigid load mounted on Navy Mediumweight Shock Machine plotted on four coordinate paper. 5.5 ft hammer drop, 3 in table travel, 4423 lb load mounted as per MIL-S-901 (after Vigness[8]).

parts. Thinking of amplitude and phase, we would then obtain, as the transform of some transient of interest, two graphs: one containing the amplitude density as a function of frequency and the other as phase angle as a function of frequency. Now, the relation between the Fourier Transform so plotted and the residual shock spectrum is as described by Morrow above. Thus the transform contains all of the information on the transient, the same information as the pulse shape itself only in a different form. The shock spectrum, however, contains half of this information, the amplitude plot, and contains it in a form convenient for approximations to the response of a linear system to the transient. The shock spectrum is a useful condensation of the information contained in the pulse itself.

This rather fundamental aspect of the shock spectrum leads the student to pursue it in the hope that it will teach him new and clever ways to look at transients and acquire improved intuition and understanding of shock response. Thus, a collection of articles have sprung up in which the authors present calculated shock and Fourier spectra for all of the typical pulses.

Gertel[26] published an article which presented shock spectra for many of the commonly used test shock pulses. He also spent a good deal of time endeavoring to show that the high frequency asymptote of the spectrum should be the maximum acceleration of the pulse, whereas the low frequency asymptote should be tht total velocity change. This type of thinking has been discussed elsewhere and still remains at least not clearly proved. Indeed, in a good deal of the shock spectra published, the frequency bandwidth presented is insufficient to make the point clear. Plainly though, Gertel believes this and tried to promote it in his article.

One of the better discussions of the shock spectrum is given by Vigness[27] where he collects many of the "points to remember" that are brought up here and there by numerous other authors. It is a very helpful article. Schell[28] has presented another detailed account of spectral characteristics. He has examined the effects on the shock spectra of slight practical variations in pulse shapes. For example, he has taken the terminal peak saw-tooth and studied the effects of a noninstantaneous drop in the pulse and also a nonlinear rise in the ramp portion. Should a study into the damaging portions of the spectrum be initiated this paper would prove quite helpful in relating experimentally realizable pulses to the mathematically ideal.

Impedance Effects

Equipment mounted at any location affects the dynamic motions at that location. This is often a severely complicating factor which often deters the designers from performing a motion history study for some proposed equipment location to specify the equipment able to survive such motions. As would be imagined, the heavier the equipment, the more this becomes a problem, and the effects are equally a problem in shock

and a problem in vibration. Were it not for these problems, we could much more easily deal with the question of the specification of dynamic tests.

With regard to shock, it appears that this fact was significantly dawning upon the pioneers in the field about 1956; but papers to date still appear to ignore it. O'Hara of NRL who has written extensively on shock response, and who has been a principal contributor to the Navy DDAM, wrote a history[29] of the theoretical development of the shock problem which describes the discovery of the foundation effects.

In the NRL writings, this is referred to as the "spectrum dip" effect. Actually the structure behaves as an absorber at its natural frequencies and absorbs the motion of the foundation causing any measured spectrum of motion at the foundation-equipment interface to decrease in amplitude at the equipment natural frequencies. Several experimental and theoretical explanations of this effect have been published.[24,30,31] The result of these considerations is that if the impedance of the foundation is not large compared to that of the equipment, the effect is severe.

Imagine a situation where a shock spectrum or vibration spectrum had been specified as a test requirement, and had been developed by measurements at some proposed equipment location without regard to the future influence of the equipment on that motion. Further imagine that the impedance of the equipment was not small with respect to that of its foundation. When the test was made some spectrum dips would occur at the equipment natural frequencies due to the absorbing effect of the equipment. The test would be considered insufficient at these frequencies, and means would be found to increase input to the equipment at these frequencies. Now these natural frequencies of the equipment are precisely the frequencies that are capable of doing the greatest damage. Thus while the equipment in service will decrease the foundation motions at these frequencies, the test with its increased inputs at these frequencies will severely overtest the equipment. Vigness[3] presents an illustration where strict adherance to motion specification would result in an excitation 30 times as large as would be encountered in the field. Thus the whole problem of how to monitor excitation becomes complex. It is clear, however, that one cannot merely monitor foundation motion without equipment in place and routinely assume that the equipment will be subject to this motion when in use.

This idea has given rise to literature which proposes that force be monitored rather than motion, as well as the comments that the vibration systems be "equalized" for a rigid mass of the same weight as the equipment and then the equipment tested without further input modification.

Blake and Belsheim[32] in their article on impedance effects in shock and vibration also explain the severe overtest that may result from neglect of foundation impedance but offer no solution. Quoting from their conclusions, "It is suggested that a survey of typical foundation and equipment impedances together with use of the theoretical

relationships summarized here, may reveal simple design and test rules." The task still remains to be done.

The Navy has avoided this problem through the use of the shock machine and the shock barge. Especially with respect to the two shock machines, is no attempt made to account for varying foundation impedance. The Navy accepts as proof of ruggedness, the equipment's ability to pass these tests, and ignores the inevitable inaccuracies. Thus it effectively accepts the test machine foundation as representative of the shipboard foundation. Naturally, this procedure must occasionally cause severe overtest and undertest. Waivers have to be prescribed for situations where the test is impossibly severe and hence certain equipment goes to sea without any test at all. Thus the present lack of an adequate method to assure proper foundation impedance contributes significantly to the overall imprecision of shock tests.

The Navy Dynamic Design Analysis Method, DDAM

Because a great number of Navy equipment such as boilers, engines, shafting, reactors and the like, are far too large for testing on any machine and could not be tested by actual explosions, the Navy has authorized a certain analysis procedure, the Dynamic Design Analysis Method,[20] which if satisfactorily carried out on designs has been accepted as proof of their shock adequacy. It is essentially the normal mode approach, with the modes combined statistically, as previously discussed, together with an ameliorating factor based upon equipment weight that attempts to allow for the spectrum dip effects. It has experienced much difficulty and has not been received with open arms by manufacturers, but it is a rational approach. The development and continued upgrading of this approach has been the continuing mission of the Structural Mechanics Group and NRL.

Hyeman[23] has presented a helpful heuristic development of the DDAM equations in which he attempts to more simply and intuitively present them. Very quickly, in a page or two, he makes a plausible argument for their validity and indeed suggests more reasonable terms than participation factor and some of the other terms in the DDAM. One would hope that his thoughts might be considered in a more understandable rigorous development of the theory. Certainly the short article is to be recommended to all making the DDAM analyses.

Bort[33] presents some interim results on experiments conducted to show the adequacy of the DDAM (basically normal mode analysis) in predicting shock loads, and it is shown quite accurate and conservative in most cases. Although some unconservative results occur, these are probably due to inaccurate modeling rather than theoretical inadequacy.

Recently an ad hoc non-Navy review committee was established and met to review Navy shock technology. In view of the considerable difficulty industry has met in satisfying DDAM requirements, the committee recommended[34] temporary suspension of the requirement until industry becomes more able to cope with it. They recommended more educational

material be prepared and extended efforts to train more qualified personnel be initiated. A guide[35] for DDAM users has been prepared to increase industry understanding which should contribute to improved dynamic analyses.

It is felt that in the DDAM lies great hope for increased understanding of shock test and design. We heartily support its increased utilization. As an illustrative outgrowth of the understanding promoted by the DDAM it is felt that since NRL has had success in specifying spectrum inputs to the DDAM, that this same reasoning can be brought to bear in specifying inputs for the experimental shock tests.

Shock Specification

The specification of a shock especially with regard to shock testing is another problem area that remains unresolved. Except for the Navy High Impact Machines, most agencies specify either a pulse shape or a shock spectrum of the motion of the equipment attachment point. Naturally, the impedance concepts, that may alter this motion, cannot be accurately accounted for. They are more or less allowed for in tolerances placed on the motion specifications. One method is to adjust the test machine to produce the desired motion with a rigid dummy mass in place of the equipment and test the equipment on the machine so adjusted, even though the presence of the more elastic equipment may somewhat alter the motion.

An interesting collection of papers each discussing shock specification has been presented at the Shock and Vibration Symposium in 1965. Kouppamaki[36] begins by arguing for use of the shock spectrum for shock specification. Vigness[37] then presents a paper arguing for a faired pulse of some shape with a 15% tolerance above and below, disregarding the hash (or high frequency component). He states that while spectrum methods and fine and good, only a few laboratories have the instrumentation to compute the spectrum from the measured pulse. He presents many photographs which show how well various testers can meet a given pulse shape specification. Ostergren[38] presents a paper in which he explains how one goes about thinking out the process of developing a pulse to meet any given shock spectrum which is quite convincing. Palmisano[39] then presents a paper in which he describes the various types of specifying and presents graphical displays of many Fourier transforms of the various traditional shock pulses, concluding that Fourier Transforms specify shock well. He introduces the Bode diagram as an additional way to record the shock. Finally Schell[40] comes up with some totally new idea called the "proximity spectrum" where he looks at the maximum relative motion of a two degree-of-freedom system, thus giving an even more complicated way to look at shock, certainly of unproved usefulness. As one might expect the group comes to no agreement concerning the best way to specify a shock.

As additional demonstration of the general lack of agreement concerning the specification of a pulse shape or a shock spectrum, McWhirter[41] moderated a 1967 panel session of renowned practitioners in this field which came to no definite conclusions. Howlett[42] presents an interesting article critical of shock spectrum specification in which he claims to have examined the response of a one, two and three degree-of-freedom system to three transients with similar spectra and found differing responses. The article is not completely clear and would have to be thoroughly studied in order to be convinced of his criticism; furthermore, his arguments did not receive any agreement from the discussion printed, and hence is not too convincing.

Since the shock spectrum is an incomplete description of any transient, one is led to expect that several transients could be developed which would produce identical shock spectra. It would be interesting to study the response of various linear and nonlinear systems to different practical pulses with identical spectra, thereby establishing some feeling for the limitations of the specification of shock by means of the shock spectrum.

An interesting effort to back calculate the waveform from the shock spectrum has been undertaken by Brooks.[43] He assumes a characteristic waveform shape and then develops its particular values from the spectrum. Therefore, one might expect that a shock spectrum together with some characteristic waveform parameter may provide an adequate shock specification means.

In summary, specification of shock by a pulse shape is favored because it is inexpensive to measure. Vigness[37] proposes that the pulse be filtered and that only its "faired" or mean value be subjected to tolerance limits which he feels should be 15%. Naturally, it is difficult to pick one pulse sufficiently severe enough to envelope a given environment. Shock spectra specification is favored because it usually only sets a minimum value of the spectra, which can easily envelope a wide variety of measured spectra, and gives results in a form capable of being easily incorporated into design through normal mode analysis. In discussions with dynamics staff at Boeing[44] it became clear that they were most impressed with the spectrum methods for this latter reason.

Damage Potential of Various Portions of the Shock Spectrum and High Frequency Shock

Let us pursue the idea of the comparative importance of the different frequency regions of the shock spectrum. Ideally we would like to be able to look at a given shock spectrum and imagine its potential for damage to equipment. If normal mode theory is not over simplifying the situation in some subtle way that we are unaware of, we are led to believe that high amplitudes in some bandwidth on the shock spectrum have greatest potential for doing damage to a structure which has a natural frequency near that bandwidth together with a corresponding mode shape that can be excited through the attachment points. This is

something of an oversimplification for clearly excitation at, say 2 Hz with an amplitude of 40×10^6 g's, applied to a structure whose first mode response was at a frequency of 100 Hz, would fail the structure very much the same as the zero frequency centrifuge is able to do. The damage potential of the high frequency protion of a shock spectrum was discussed during an initial meeting between NCEL and NELC that was held to orient the study. A separate publication[45] has reported those results which show stress proportional to modal velocity regardless of frequency. It is hoped that such practical results can continue and that they can be used to prescribe the final improved dynamic test.

Before specifically discussing the slight available other literature on this topic, let us bring up some shock spectrum plots for different locations on a ship that were presented by Keil;[13] note Figure 1 again. One would expect that shock as felt at the keel to be most severe, while that suffered on the main deck and superstructure to be considerably filtered and attenuated with less resulting damage potential. Now note the differences in these two spectra. The keel sees higher values for frequencies in excess of 30 cps, and reduced values below 20 cps. Note also that the velocity spectrum is a good deal flatter than the acceleration spectrum, a fact we shall come back to later. Though there is probably a good deal of information lurking here, we shall content ourselves with noting that of the three displacement time curves in the upper portion of his figure, the one on the keel does indeed look most severe. The only resulting difference we can see in the spectra that would lead us to believe the keel spectrum is the most severe, however, is the fact that the keel spectrum has increased high frequency content at least from 80 cps on down.

Now let us consider high frequency shock, i.e., shocks with appreciable spectrum content at frequencies in the thousands of cycles region. Hughes[46] has made measurements on "Gun Setback Acceleration," a confusing title; it is assumed this means the actual acceleration of the projectile, and so the actual measurements here are not significant to equipment, but it is interesting to note the extreme high frequency content he has measured. For example, he is recording 30,000 g's at 10,000 cps, and 18,000 g's at 20,000 cps, as shown in Figure 4, taken from reference 85.

Britton[47] presents data on pyrotechnic shock (in this case reentry vehicle shock when subjected to dynamic inputs from linear shaped charges, explosive bolts, etc.). The acceleration spectra, he reports, generally increase with frequency from about 200 cps to 5000 cps, the values running from 10 g's on up to almost 10,000, for example see Figure 5. Although many of his spectra show fall offs in amplitude at 5000 cps, it does not seem to this author that he has in anyway shown an upper frequency limit to extremely high g levels. It would seem that if his analysis could be carried to higher frequencies he would have found still higher g levels. One point that we will again return to, but that is conveniently brought up at this juncture in connection with this paper, is the graph of Figure 5 taken from reference 47. The values extent approximately from 7000 g's at 4000 cps to 100 g's at 100 cps.

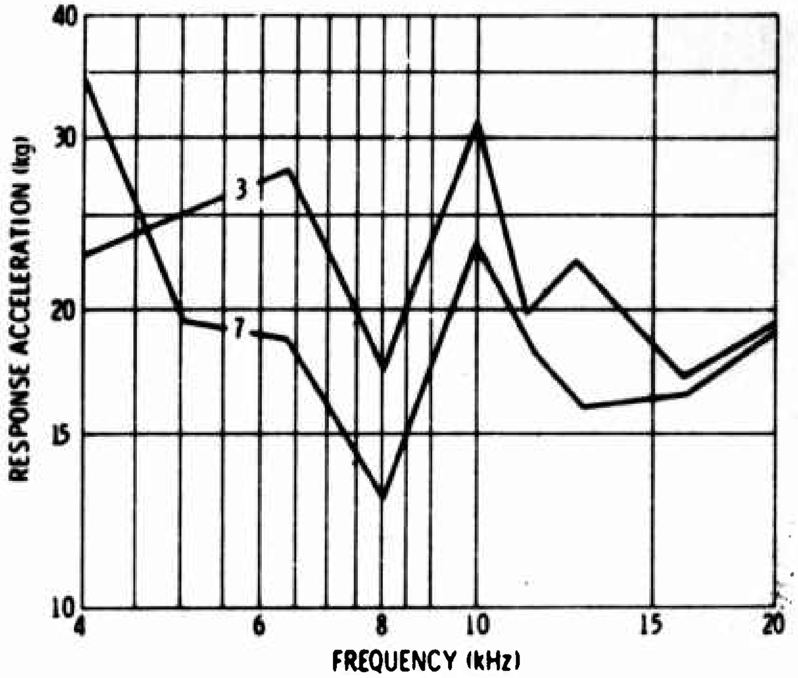

Figure 4. Response spectra of partial 5"/54 gun shock, computed at 2 percent critical damping on analog computer (after Hughes[46]).

The difference in level between 100 g's and 7000 g's poses severe problems in instrumentation. Recalling that velocity is acceleration divided by frequency, note that in this case and in the vast majority of similar cases, had velocity been plotted rather than acceleration, the magnitudes throughout the spectrum would have all been of the same order, thus posing a much reduced measurement problem.

Barnett[48] has presented data on spacecraft separation shocks which show shock spectra to 7000 cps; in almost all cases the acceleration levels are rising to the 7000 cps frequency and indicate no fall off in amplitude with increasing frequency. He shows values of the order of 2000 g'd at 7000 cps. This article prompted appreciable discussion indicating concern over this high frequency from the attendees of the conference. Gertel, discussing Mr. Barnett's article,[48] made an interesting comment which indicated that as late as 1968 he did not expect to see high acceleration levels for the high frequencies.

A similar description of high frequency shocks performed at North American Aviation on explosive spacecraft separation[49] shows shock values in excess of 10,000 g's in the range of about 7000 cps. Both of these authors comment quite emphatically that no failures were encountered even though the shock levels were quite high.

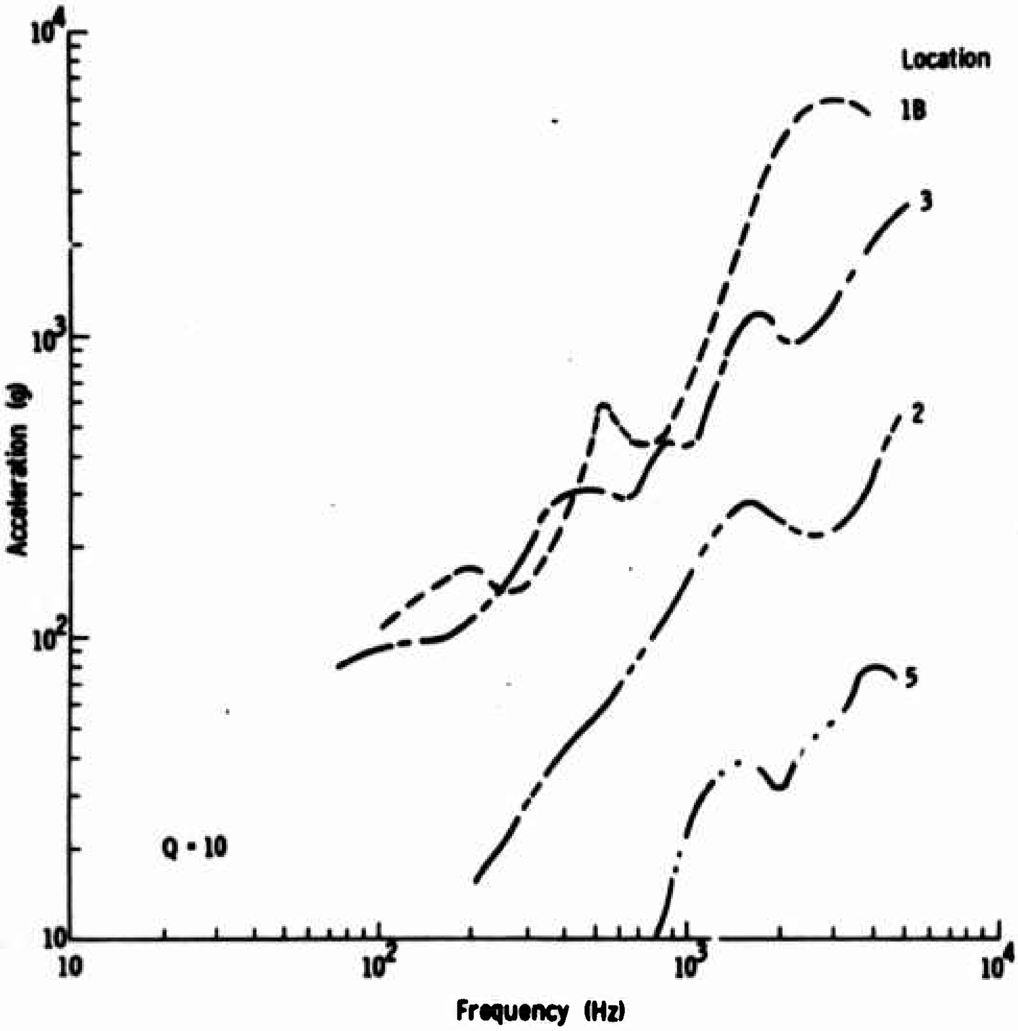

Figure 5. Drogue disconnect tests shock response spectra envelopes for original test configuration (after Britton[47]).

Blake[50] presents an article in which he expresses concern over the problems of simulating high frequency shock in the range of 500-5000 cps, and cites the problem that the wavelengths become short and transducer location is very critical. He more or less leaves the problem and just cites his difficulty with it; his concern does lead one to conclude that high frequency shock is a problem and should not be quickly dispensed with. Finally, in conversations with Boeing[44] concern over high frequency shock was expressed. They were convinced that several of their failures were due to frequency content in the thousands Hz range.

Thus much evidence exists that, at least in shocks due to explosions, much very high frequency content can be found. No one, however, presents specific data on failure due to these high frequency content shocks, although some have been verbally reported.

Gertel[51] presented a paper in which he attempted to show that high frequency content need not be considered. The work is not convincing and has had no follow-up in the literature. He considers beams by simple beam theory and shows that if the modes have the same maximum acceleration, the higher frequency modes show appreciably less stress. As was pointed out in the previous paragraphs, constant acceleration does not appear to be representative of the situations in which high frequency shock arises. Further as frequency increases modes become extremely complicated with many nodes and antinodes hence the modes loaded by these high frequencies will be generally well beyond our capacity to model or instrument for measurement. Also as frequency in any given beam or structure increases, at some point the vibrations become shear waves, plane waves, surface waves and the like, traveling throughout the structure. Gertel in no way attempted to explain these effects. His work is thought of as incomplete although a worthy effort in a worthwhile direction.

It is expected that the only situations in which one will be able to discard the effects of high frequency shock will be where the environment can be studied and found lacking in significant high frequency content.

On the Navy lightweight shock machine which interposes little or no fixturing between the equipment and the anvil, one would expect considerable high frequency content. Note Figure 6 taken from Vigness, reference 8, which shows 1800 g's at 900 cps for the lightweight machine and compare it with Figure 3 for the mediumweight machine. On the mediumweight machine the procedure for which specifically describes a stack-up of channels with a 60 cps natural frequency, one might expect the high frequency to be considerably attenuated or filtered, after that frequency which is shown in Figure 3. Barge tested heavyweight equipment can be tested with little or no structure between the water born shock wave and the equipment, thus it may experience severe high frequency content. One would expect equipment, mounted stiffly to ship structure in intimate contact with the hull plating which receives the water shock, could potentially see large high frequency content. Deck mounted or other structure mounted remotely from the hull plating would similarly be expected to see little of the high frequencies.

Forkois[52] in general terms speaks against the measurement of the high frequency shock components. He recommends that the instrumentation system be filtered so that these more complicated aspects remain unnoticed. One can envision considerable research into this area alone. For example, consider a simple analyzable structure tested to failure with a very broad banded shock pulse (one with nearly constant frequency content). The filtering out of certain portions of the broad banded shock pulse could determine the contribution of the different frequency ranges of the spectrum to damage. It would seem that such a test could be carried out with small structures on an electro-dynamic shaker-shock setup; the equalizing filters could eliminate certain frequency bands in the resultant pulse. It could probably also be accomplished with a

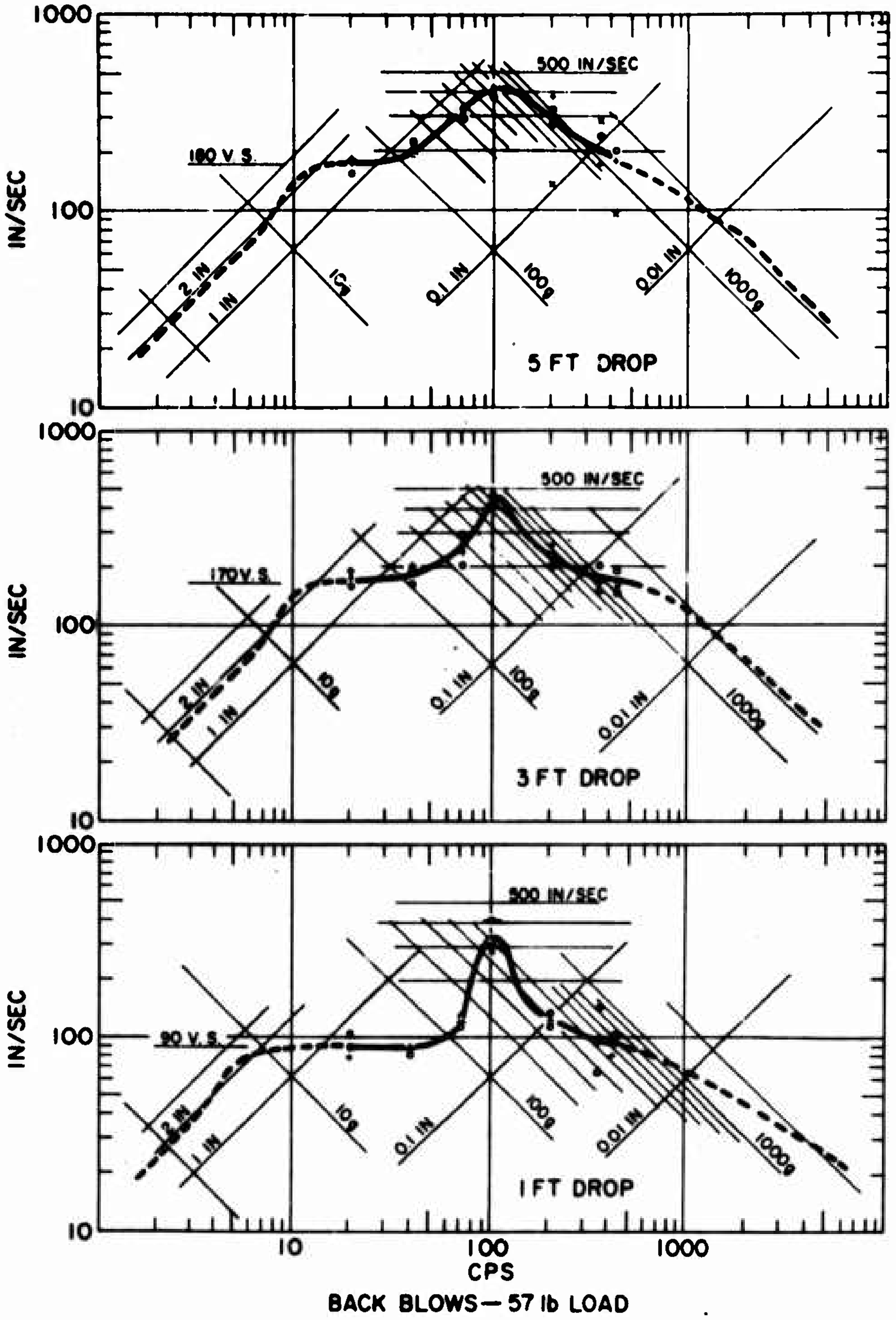

Figure 6. Shock spectra for motions of a rigid mass of 57 lb attached to the 4A plate of the HI Shock Machine for Lightweight Equipment. An equivalent velocity-shock (V.S.) for frequencies between 10 and 40 cps is indicated (after Vigness[8]).

hammer by interposing pads of various materials to filter out or attenuate certain frequency bandwidths. Such as experiment would certainly add to our knowledge of shock damage.

As attempt has been made to use statistical vibration theory for the prediction of shock response. In their article Manning and Kyung[53] briefly describe one experiment in which the experimenters were able to predict the order of magnitude of the response and its general form by use of the statistical formulas. The author has discussed the potential for these efforts with Dr. T. D. Scharton of Bolt, Beranek and Newman in Van Nuys, California,[54] a colleague and associate of Jerome Manning. Manning and Kyung's paper is the last published work on this topic; Scharton and Manning plan to cooperate in a joint effort to pursue the uses and conclusions of statistical vibration theory to the general shock testing problem. It appears that their contribution will be in the area of high frequency shock,where many modes interplay and transmit the energy along. Questioned as to whether they have done any thinking along the lines of the type of damage that will be associated with these high frequency components, Scharton replied that they have been more or less concerned with merely simulating what has been measured in the pyrotechnic shock environment rather than looking into failures.

In summary, little appears to be known about the damage potential of the various regions of the shock spectrum. Many people are concerned about high frequency shock and have measured it. Little data exists to cause anyone to believe that high frequency shock does in fact cause any high percentage of failures.

Effect of Fixtures

A fixture in general alters the dynamic environment transmitted though it. While this could be no problem at all, if input monitoring was specified at the equipment fixture interface, it is a severe test alteration if monitoring is accomplished at the machine fixture interface or as in the case of the Navy shock tests, no monitoring accompanies the change. The obvious resolution of the problem is to specify monitoring at the fixture equipment interface.

In the case of the floating shock platform heavyweight shock tests[5] the mounting is specified as, ". . . the equipment shall be installed in a manner to simulate the most severe condition likely to be encountered." To illustrate the spirit in which this is interpreted: Schrader[55] of the Navy has presented an article where he is explaining to the industry that they should build low natural frequency fixtures even though they are expensive because they will ameliorate the damage potential of the N , High Impact Floating Shock Platform. Figure 7 taken from that reference shows the deck response of the floating shock platform (FSP) as compared to the response of his equipment on a flexible fixture; note the severe alteration of the shock by the fixture. It will also be noted that substantial high frequency hash is seen on the velocity trace of the deck compared to the type of hash seen on the

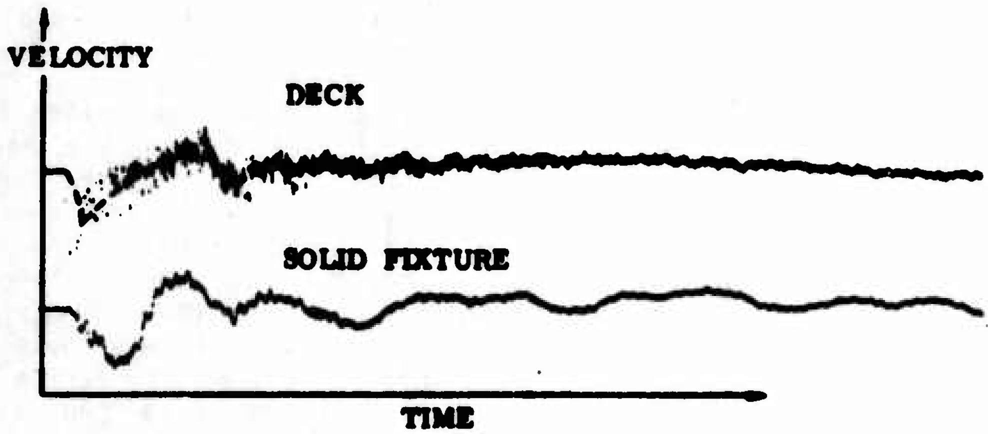

Figure 7. Floating shock platform deck response and flexible fixture response illustrating severe effect of fixtures (after Schrader[55]).

equipment. Schrader also presents the fact that the fixture should have been the same fundamental frequency as is to be expected on the ship but does more or less warn that the higher the natural frequency of the fixture the greater will be the g level seen by the equipment.

Incidental to fixtures and with further reference to the point of the last section, in printed discussion of reference 55, Bort notices that the frequencies and the accelerations are approximately equal in the magnitude for all the different parts with differing natural frequencies. This is to say that the velocity spectrum is about constant, and adds credence to the comment made previously concerning measurement of velocity rather than acceleration.

In summary, it is concluded that fixturing is, has been, and will remain a troublesome area. If a rationale is found to determine precisely what must be delivered to the test item, then monitoring at the equipment-fixture interface will assure an adequate test. When proposed foundation impedance can be determined the shock can be delivered through a fixture simulating this impedance. Otherwise ambiguity exists. A kind of an unimpeachable theorem underlies all test credibility arguments, which can be invoked at anytime, and that is as the test more nearly duplicates the field conditions it becomes more reliable. Thus a fixture can be justified by showing its similarity to the actual foundation of the equipment in the field.

Shock Measurement

The measurement problem might well go unnoticed depending on the literature one happens to read. Virtually no mention of instrumentation problems ever appears in the aircraft and missile test literature. Shock acceleration spectra are reported routinely and often. However, there

appears to be significant problems and they seem to stem from the piezoelectric and resistive accelerometers. These are low frequency limited devices with very high natural frequencies but undamped. If as in the Navy one wants to integrate the acceleration back to displacement to, for example, confirm what has been observed about gross motion by a camera, the absence of the low frequency information prevents this. Further if one is experiencing a shock that contains appreciable high frequency as high as the transducer natural frequency, the transducer will response resonantly giving an untrue picture. Finally, as has been discussed, acceleration amplitudes vary over many orders of magnitude with frequency, so that if one measures levels at high frequency, the amplitudes are go great that they obscure the levels at the lower frequencies. In a large measure the Navy has eliminated these problems by using velocity transducers which are unable to read the high frequency, and are not troubled by the large range of amplitude that is found in acceleration measurements. It appears to this author that effort should be expended to find velocity transducers with wide frequency bandwidths, which could potentially reduce at least those problems associated with the large change in amplitude with frequency.

Clements[12] has documented zero shift and other problems associated with the use of piezoelectric accelerometers for high frequency shock measurements. Oleson[10,11] has written on the difficulties of shock instrumentation and describes the rather large velocity transducers that have been used in shipboard shock measurements. He also describes some special mountings he has developed for damping out the high frequency input to piezoelectric accelerometers. Results of this work are disturbing in the light of the extensive shock acceleration data continually taken in the aircraft and missile fields with never a mention of instrumentation problem. One can only hope for the possibility that perhaps in the one case the low frequency components are needed quite accurately to make meaningful integrations, whereas in the other case where equipment natural frequencies are high and the accelerations at higher frequencies are sufficient to the needs, and both groups are being accurate.

DeVost[56] presents an article where he finds transverse response of accelerometers as great as 65% of the transverse input (the response should be zero). It is his opinion that the high frequencies cause these errors, thus in the high frequency shock reported in the section on high frequency shock, the question arises as to what portion of what has been measured might be due to transverse motions. At least information is available to make one skeptical of instrumentation technology with respect to shock, and this will certainly affect any attempts to specify monitoring.

The Seriousness of the Shock Problem

In conclusion it is felt that shock and shock testing are still serious problem areas which cause an undue proportion of equipment

unreliability. The severe inflexible tests we now use can undertest and overtest. Shock failures are obscured because they occur only in near disaster situations and usually long after the equipment was tested. There is little chance of anyone being assigned responsibility for the failures.

In spite of the difficulty in specifying, deliverying, and even measuring a shock input, some rational approach must be developed by which logically arrived at input must be delivered to the equipment which must then show itself adequately designed to withstand the shock.

The Cycle of Healthy Shock Reliability of Equipment

The shock reliability problem is a kind of a circular one; a whole cycle of effort must be introduced. The first step will be to convince the Navy that considerably increased concern over shock is needed. Then realistic loads must be specified, very probably by providing a shock spectra together with a rational amelioration scheme to take care of the situation where impedance effects can cause overtest. Next an accurate test must be developed which beyond any shadow of a doubt does indeed deliver these loads to equipment so that a manufacturer will definitely believe that his equipment will be tested to these loads. The manufacturers and their designers will have to be taught how to anticipate the effects of these loads on their equipment so that they can design sufficiency into their equipment. No manufacturer desires not to comply with a specification. If they knew in any way how to design for the magic Navy Machines they would not hesitate to do so. Perhaps the Navy needs an educational team that could go around and teach these design methods to the manufacturers. Schedules will have to initially be written into the contracts to assure that the manufacturers are designing shock resistance into the equipment. Review by capable personnel must be built into the design process until such time that adequate shock design becomes common knowledge. It is doubtful that this kind of training can be expected from the colleges for several years. They are certainly not aware that it is needed at this time.

Thus around this cycle one can see the areas of development needed. Monitoring of test is essential to convince and prove that the specified loads are indeed being delivered. To monitor is to measure and hence the measurement problem must be at least temporarily solved to the extent of the state-of-the-art, at present, and then better as improved methods are developed. The theory must be crystalized and taught. The test that does deliver the specified environment must be developed. The amelioration scheme for low impedance foundations must be developed; probably initially as was done with the DDAM. The test alone which is the main concern of this research is truly only a portion of the total problem which will have to settled before significant improvement in shock reliability will be attained.

VIBRATION

The vibration environment includes the long term continuing dynamic field in which the equipment must always operate together with some provision for temporary increased levels due to aggravating circumstances - severe weather, temporary machinery unbalance, and other like situations which might cause temporary increased dynamic fields. Equipment attached to a structure containing this environment responds dynamically to this environment in such a way that each particle of its mass undergoes oscillatory motion; the accompanying accelerations give rise to inertial or D'Alembert forces which load the equipment.

The type of failures that occur as a result of vibration are two-fold. When a lightly damped structure is excited by a periodic force with appreciable harmonic content at one of the natural frequencies of the structure, the oscillatory amplitudes increase severely. Thus a failure mechanism can be anticipated that is due to build up at resonant frequency to so great an extent that either deflection limits or elastic limits are exceeded, resulting in a failure. The oscillatory amplitudes may become so great that the failures will no longer be fatigue failures but rather high excursion failures, though in some cases these could be considered low cycle fatigue. In general these failures are not sensitive to the accumulation of a large number of cycles but rather due to a momentary excitation at the appropriate level and frequency. Conversely, the second failure mechanism is fatigue. Comparatively lower amplitudes with stresses great enough to cause fatigue damage to accumulate, persist over a longer period of time. Suddenly, after many many cycles and without appreciable warning, a fracture occurs. An endurance limit, a stress below which fatigue damage does not accumulate, is generally believed to exist and its value is taken to be the highest reversed bending stress for which 10^6 - 10^7 cycles can be accumulated without failure. Thus theory suggests that if at each modal frequency the number of cycles that define the endurance limit can be tolerated by the equipment, then the endurance limit is nowhere exceeded and the equipment can stand that level oscillatory excitement indefinitely.

It must be cautioned that the above discussion pertains to stress induced mechanical failures. Many other failures can occur such as relay chatter, misalignment, electrical modulation, etc. Current theory cannot anticipate any long term cumulative effects that can be used in rationalizing test duration. We must at present presume that a test sufficient to cause mechanical failure will also precipitate the non-mechanical failures and certainly caution that careful monitoring of equipment total function must be incorporated in all vibration testing.

Types of Vibration Environment

For purposes of this discussion, we shall think of the vibration as a single signal. In reality it can be vastly more complex, ultimately with each point in the structure undergoing three dimensional motion to

a large degree even independently of the other points of the structure. But even this single signal interpretation of a vibration warrants some thought.

The signal may be periodic or nonperiodic. Strictly periodic definitely implies that the signal is composed of harmonic sinusoids, i.e., all sinusoids present in the signal have frequencies which are integral multiples of the fundamental frequency. Each constituent sinusoid must repeat itself identically in each fundamental period. This can only occur if the frequency of the sinusoid is an integral multiple of the fundamental frequency. Also, everything in the signal except for any DC component is included in the sum of this set of harmonic sinusoids. The spectrum of such a signal is said to be a line spectrum since a definite set of distinct sinusoids make up the complete signal. If one were to plot the harmonic content as a function of frequency, the plot would be a series of lines at the distinct frequencies, all multiples of the fundamental.

Nonperiodic signals may be deterministic or random in nature or some combination of both. Deterministic here means a combination of nonharmonic sinusoids; if the signal is the sum of several sinusoids whose frequencies are not integral multiples of some fundamental frequency then the signal is definitely deterministic but not periodic. It will have a line spectrum representation of frequency content but the lines will not be harmonically related. Random signals, finally will have a continuous frequency spectrum, their characteristics are only identified statistically, and involve a further test for stationarity; i.e., their statistical characteristics do not vary with time.

It is expected that a frequency or spectral analysis of some vibration record of sufficient length would classify it according to the above. Bendat, et al[57] have discussed a proposed standard for organizing such analysis and referenced computer programs that automate the required calculations. As described above such an analysis should yield a harmonically related line spectrum for deterministic nonperiodic data and finally a continuous spectrum for random signals. The signal, if found random, would then have to be analyzed for stationarity to find to what extent noise theory applies to it. Smith[58] describes instrumentation to accomplish this and translate the result to test specifications.

One has to expect that no definite answer would come back but rather the signal would appear to some extent composed of this portion of one and that portion of the other. Mature engineering judgment would have to be applied to logically resolve a signal into a testable specification.

Types of Vibration Tests

There now exist four different catagories of vibration tests according to the way in which the different frequencies are introduced; there are sinusoidal search and dwell, swept sinusoidal, broad band random and swept narrow band random.

Vibration testing by resonance search and dwell has been a primary vibration test for many years now and is currently specified by the Navy for shipboard equipment in MIL-STD-167,[6] and also in a 1964 USAF Aerospace specification, MIL-STD-810 (USAF).[59] The test generally involves a certain amount of time during which a low level vibration is applied to the equipment with varying frequency to search and find the "significant resonances." Then vibration is applied at these significant frequencies for extended periods of time to see if failure occurs.

Such a test procedure has many disadvantages and hence it is the opinion of this and many other authors that the test is obsolete and should be discarded. The main fault of the test is that all significant resonances cannot be found. The natural tendency would be to select those resonances causing large obvious structural deflections and those producing audible sounds, which may or may not correspond to potentially damaging resonances. It has to be clear that when equipment becomes complex with several subassemblies, such as vacuum tubes, many of the potentially damaging resonances will be missed by any given test engineer no matter how clever and perceptive. Thus in this type of test, equipment response at those unnoticed frequencies passes untested. Vigness, in his chapter in Crandall,[3] comments with regard to equipment used in jet aircraft (where the environment has been proven decidedly random) "The emphasis placed on sinusoidal tests, as shown in Figure 8.19 for equipments to be used in jet aircraft, is regrettable; . . . The high rate of failure of electronic equipment in military aircraft can, in part, be attributed to the unconservative factors associated with representing random vibration by sinusoids." Soboleski[60] also mentions the probability of missing the significant resonances and presents a study of the actual experimental accumulation of fatigue damage during dwell testing. Gertel[61] in an extensive article on vibration test elaborates many of the disadvantages. Wignot[62] notes that the test fixture will influence the resonant frequencies; thus the resonant frequencies and mode shapes stressed during test are different from those the equipment will see in actual service. Morrow[4] also summarizes these difficulties, but does mention that the test has value when the environment to simulated clearly has a single established frequency. Bangs,[63] conversely speaks in favor of the search and dwell procedure by noting that it locates the mode shapes. He feels that this method is useful at lower frequencies and inexpensive. This author disagrees with Bangs' contentions. Firstly, the mode shapes are not essential in qualification or proof testing; since most electrodynamic equipment cannot reach very low frequencies at any amplitudes of consequence, the usefulness of the method at low frequency is questionable. Finally, it is doubtful if the slow sine sweep test is more expensive than the search and dwell technique.

In a search and dwell test only those mode shapes that are deemed to be resonances are significantly tested. It is highly probable that the weak link in the fatigue chain will lie in some mode that is not dwelled upon and hence goes on to service without test. Single frequency

dwelling can only be justified when this is known to be the environment and the only environment. If any number of single frequencies may be applied to the equipment, then surely it must be tested against each, and not only those that are chosen arbitrarily as significant.

It is felt that the traditional resonant search and dwell technique currently specified is inadequate. The simple fact that most of the potentially damaging resonances may pass by untested, is certainly reason enough to discontinue the procedure no matter how long it has been in use nor how reliable equipment so tested has proved. Where the tests under consideration here are routine and applied to wide varieties of equipment due to operate in many different conditions, no argument for single frequency testing can be made plausible. The other procedures, slow sine sweep, narrow band random sweep, and wide band random testing all are without this potential drawback.

It will be noted that abandoning of this procedure would eliminate the problem of "test engineers establishing which resonances should be tested" as well as the problems encountered in tracking the resonant dwell frequency.

The slow sine sweep vibration test, just as the title implies, is a test in which sinusoidal excitation is applied to the equipment with the frequency slowly varying over the anticipated environmental bandwidth. The sweeps are run in both directions, up and down so that nonlinearities with jump phenomena equally excited. The sweep rate is set such that each and every resonance is excited for either equal time periods or equal numbers of cycles. Granik[64] and Gertel[61] both discuss calculation of sweep rates for this type of test. Granik suggests a sweep rate such that each resonance is tested as severely as it would be in the search and dwell technique; this seems quite unrealistic and such that it would involve huge quantities of time, but further study would make this clear. An article on sweep rates will soon be submitted. The main point to be appreciated in this type of testing is that each resonance is excited equally without the bias of a test engineer selecting which resonance to test. Since each resonance is excited separately the equipment costs are minimal; the test merely requires a change of procedure on existing equipment.

Broad band random vibration testing excites all resonances simultaneously and hence requires much greater energy input to the specimen, but since all resonances are being excited simultaneously, the test is accomplished in a much shorter period of time. Apart from the obvious time economy in such testing, one is forced to decide if the environment in whick one seeks reliability is more nearly discrete sinusoidal or random in nature. Buchman and Tukerman[65] present a detailed summary of the data recording procedure in use at NSRDC and comment on the extreme variability of the data which does indeed support the idea of the random test being more applicable.

Booth, in his chapter on "Vibration Generation" in Crandall[3] presents a fairly complete account of the various standard tests and their associated equipment. He specifically dwells upon the broad band random and swept narrow band random tests and the problems associated with each. Since the swept random test requires about one-third the power rating of broad band test, it is, perhaps, more applicable to the heavier equipment with respect to any one vibration system. The purported advantage of the swept narrow band random test over the slow sine sweep test is somewhat nebulous.

Finally with regard to the decision of which type of vibration test to use for establishment of reliability of equipment, one would most naturally seek that test most nearly approximating the expected environment. Without sound reason or proof it is to be expected that most environments are random to some degree. Even though the main excitation of the oscillations (caused by the ships and propellers, various rotating machinery, compressors, generators, pumps and the like) is indeed periodic in nature, the actual summation of these effects coming to the equipment attachment points through paths of differing lengths and with differing degrees of nonlinearity, will be best described as a random process, perhaps somewhat limited in bandwidth due to the filtering effects of the intervening structure. Thus the most realistic vibration test will probably be found to be some form of random vibration. In order to compare the effects of the various tests that are available, consideration is now given to the various equivalence theories.

Equivalence Theories

Two types of vibration equivalence must be considered. The equivalence between the various types of vibration testing and the equivalence between any of them and actual field conditions. Root[66] is currently preparing a monograph on this subject and has reported that in his initial research he finds a great deal of ambiguity.

Many studies have been presented concerning the equivalence between various controlled laboratory tests in which fatigue is the failure mechanism. Since the random vibration test is a sequence of cycles at differing amplitudes, any assessment of the effects of the test involve a fatigue damage theory, a theory that relates the damaging effects of stress cycles at differing amplitudes. Root[67] has presented an analysis of equivalence in which he used the comparatively simple hypothesis postulated by Minor that considers a linear relationship between stress and damage. Clevenson and Steiner,[68] however, presented results of random fatigue testing which indicated this simple linear damage theory unconservative. Crede's chapter in Crandall[3] gives a development of several equivalence theories utilizing a number of different damage theories. None of the methods have been proved accurate.

One might expect, at least with regard to vibration tests which ultimately lead to fatigue failures, that given a "true damage accumulation" theory one could merely count the progressive accumulation in any

specified test and relate it to any other specified case. Such is not the case. Evidence suggests the damage depends even upon the order in which the higher stresses are applied. Further, amplitude increase at resonance due to sinusoidal excitation is proportional to a so called "Q" of the system, while broad band random excitation causes the resonances respond proportional to the square root of the "Q's." Thus for systems with different Q's at different frequencies equivalence will be most difficult to obtain (see Crandall,[3] page 223).

Another aspect of equivalence, which applies to both laboratory and field testing, is the problem of equivalence between tests at differing amplitudes. Clearly one must be able to accelerate a life test when estimated life is longer than the time that can be allocated to testing. Gertel[61] is one of the few authors who attempts to answer this problem; he presumes knowledge of the fatigue curve for the material and estimates an exaggeration factor. Strictly speaking, equivalence will only be attained for very simple vibrating systems and when one is dealing with a material that exhibits a predictable damage criteria and predictable variation of Q with frequency, it is hardly possible that for any complex system the different levels if vibration testing will be equivalent in any strict meaning.

With regard to equivalence between laboratory simulation of vibrations and actual field vibrations, again little solid theory exists. Vigness in his chapter in Crandall[3] discusses the problem and cautions away from equivalence theories by advocating tests that attempt to simulate what has actually been measured of the environment. Spang[69] gives a complicated but at least organized method of harmonically analyzing vibration records to determine if they are random, stationary, periodic and the like. Robson and Roberts[70] present illustrations of theoretical simulation of a given measured random spectral density. Actually this is the general problem of enveloping, or from data plotted spectrally selecting some percentage of the maximum amplitudes to provide an adequately reliable test. The whole of Vigness' chapter in Crandall[3] is attempting to indicate the procedures. Morrow[4] also spends considerable time on this problem. When the measured environment is low enough so that complete enveloping of all maxima can be accomplished without undue or impossible requirements being placed upon the equipment, the procedure seems logical and straightforward. When measured environments are comparatively high, however, the engineers develop logics that permit reduction in test levels below this value, and then the procedure becomes considerably less convincing.

A hope now exists for the experimental determination of various types of equivalence in the S/N Fatigue Gage (William T. Bean Associates; Detroit, Michigan, and Micro-Measurements, Inc.; Romulus, Michigan). This is a gage quite similar to an electric resistance strain gage which is reported to gradually increase in resistance in proportion to the fatigue damage accumulated; thus a measurement of the gage resistance is reported to be a direct measurement of the damage accumulated. This gage should therefore be able to record the rate of damage accumulation

in the field and during a laboratory dynamic test, thereby indicating equivalence. The gage should be equally applicable for comparing tests with different levels and durations or for the determination of equivalence between the various sine and random tests. Cost[71] has reported on one such study and concludes that the S/N Fatigue Gage should be widely applied to determine equivalence between the various tests. Certainly to the extent that the gage does indicate the types of damage due to shock and vibration environments, we are now in a position to experimentally resolve the problems of equivalence.

Purposes of Testing at Various Levels

At least four different motives for conducting vibration tests become evident. A very low level sinusoidal vibration sweep is often performed to identify and survey the mode shapes and natural frequencies merely as a design aid or for better understanding of the dynamic response of a prototype. Oftentimes it is also desired to life test an item at some level. The tests that concern us here, however, are those that enable one to assess an equipment's reliability in some vibration environment. One can imagine that a proposed design of an equipment should be first extensively tested to prove that the design, when properly fabricated will indeed survive the long term cumulative effects of the anticipated environment. Once the design is so proved then a much shorter acceptance test could be routinely used for quality control on each succeeding unit built. A discussion of the design qualification and routine acceptance tests are contained in an article by Hasslacher.[72] He comments that vibration testing should not attempt to accomplish a fatigue test for routine acceptance, but rather it should detect manufacturing errors. The design qualification test that accepts a design as adequate would be a test at levels and durations that would cause fatigue failures which would be inherent in poor design. The routine acceptance test would be of much lesser duration to locate any loosenesses or poor fastenings, cracked castings, etc. He also comments on the fact that one must assess the damage that has been done to a piece of equipment as a result of dynamic tests. He indicates that if the equipment has passed the long duration design qualification tests, it should not be considered adequate any longer because of the damage that may have been inflicted upon it. He attempts to quantify equipment degradation due to vibration tests, and present this as another cost of test.

Thus, it seems that for large production runs of a particular equipment, two tests are advisable: design qualification and routine acceptance. When only one equipment is to be built, we have a more difficult problem; following a life test the equipment may be degraded due to accumulated damage.

Statistical Vibration Theory

Statistical vibration theory[73] is a comparatively new idea which has not yet been completely evaluated or exploited. Most of its proponents seem to work for Bolt, Beranek and Newman, Inc., a fact which somewhat detracts from its credibility. The theory apparently applies to high frequency vibration where hundreds of modes are present in the resulting response; apparently, then, one need only speak statistically about the amplitude of any one mode. Since the theory is more concerned with higher frequency vibrations it is doubted that it will have much application in this study. However, it might well be anticipated that in a structure as large as a ship hundreds of modes exist well below fifty cycles, and so the theory may find applicability for ships as a whole. An interesting comment by Franken[74] is that beyond 200 cps the mode shapes become so complicated and multiple that three dimensional testing may become unnecessary because all the modes may be excited sufficiently by excitation in a single direction.

Impedance Effects and Input Monitoring

As the mass of the equipment increases with respect to the mass of the foundation exciting it, the equipment begins to effect the motion of the foundation, whether this foundation is the ship structure or the vibration machine. Actually any structure oscillating in any of its modes contains an amount of energy continually changing its form from stored elastic at the extreme amplitude positions to kinetic at the mean position. In any mode at any steady state amplitude, damping or dissipation mechanisms continually dissipate energy and hence a steady flow of energy is required from the excitation source. In large heavy equipments these energies are large with respect to the modal energy contained in some mode of the ship, thus the ship cannot deliver sufficient energy to the equipment to cause it to oscillate as it would if it were excited by the motion of the undisturbed attachment points. Thus to subject an equipment to a motion measured when the equipment is not in place may subject it to a severe overtest. Attention is also called to the fact that the equipment will reduce foundation amplitudes most drastically at its own natural frequencies; thus should the test engineer attempt to compensate for the amplitude decrease he would be increasing unrealistically the excitation of the equipment precisely at its most susceptable frequencies.

Vigness[75] devotes an article to the subject of the potential overtest that may result from demanding equipment mounting points to undergo motions that have been measured at proposed attachment points on structures. He cites an example in which overtest by a factor of 30 occurred. Finally he makes some recommendations to the effect that (a) the vibration machine should be equalized with a dead mass in place and then the test run on the actual equipment with no further equalization; (b) the ". . .vibration amplitudes of the foundation be determined together with the mechanical

impedance of the foundation and the mounted item as viewed by the foundation. This procedure. . .appears to be impractical for general field vibrations studies."; (c) a statistical study of both equipment motion and foundation motions such that a set of qualifications can be developed whereby input may be reduced when equipment response becomes unrealistically large. Only the first of his methods appears easily implementable and even it would require study to show that it offers a realistic test. The impedance problem is indeed difficult and in need of much further study.

Furthermore, the impedance problem is even more complicated than explained above. The effective mass of the equipment and the foundation changes with frequency; as the frequency increases, effective mass decreases and even massive foundations may effectively become light. Also much has been written by people concerned with aircraft and missile vibrations concerning the monitoring of equipment vibration at each of its several attachment points. Articles and comments appear to indicate that one should monitor motion according to the largest motion of any of the attachment points, and according to the smallest motion. As frequencies increase, the wave lengths or distance between nodes on the equipment becomes small and in general each attachment point will have different motion. Thus the ascribing of uniform input motion to the structure becomes incorrect, and even further the correct approach is not at all clear. Fortunately, the upper frequency limits of shipboard vibrations are quite low so these particularly complex problems will probably not be of concern.

With regard to attempts to cope with these problems, Otts[76] presents a discussion in which he proposes to control force rather than motion in random vibration testing. The idea is difficult to put in practice since he proposes to reproduce the force measured with the equipment in place on the structure, in effect requiring the equipment to be installed aboard the ship to determine the vibration test specification. Painter[77] describes a test where measured environmental accelerations were obtained with and without an equipment in place. The monitoring was then specified to be according to the acceleration envelope measured without the equipment in place except at equipment resonances. At a resonance the acceleration was limited such that the force developed between the shaker and the equipment did not exceed the maximum force measured between the equipment and its foundation during the original experiment. He claims this procedure largely eliminated the "high levels of overtesting introduced by the conventional approach."

Fixtures

If the equipment can affect the motion of a shaker or its actual supporting structure when installed for service, it can similarly affect and be affected by the fixture with which it is attached to the vibration exciter. Painter[78] also presents an article in which he discusses problems he has encountered in fixturing and comments that in most

situations the fixture effect simply cannot be compensated for; distortion is always present. Here, however, he is testing at frequencies a good deal higher than we are concerned with. In a discussion with Reece of Wyle Laboratories,[79] we were told of a company whose main business comes from designing dynamic fixtures to be used in equipment test. It was reported that this company has the reputation of being able to design a fixture such that most any equipment will appear to have passed the test. Certainly any fixturing effects could be cancelled by requiring monitoring at the fixture equipment interface, a practice we certainly support.

Scharton[54] has developed an interesting approach to the fixture problem, especially as encountered in the very high frequency ranges. He attempts to make very flexible fixtures with hundreds of modes in the frequency bandwidth of the test. He reasons that a structure only passes frequencies lower than its first mode frequency and then selectively at each of the rest of its natural frequencies. Since normal structure also contains hundreds of modes, especially when one considers the higher, even acoustic, frequencies, he reasons that such a fixture will better simulate actual environments. Evidently he has met with some success in this research for he has convinced JPL to order some of these fixtures for study.

Final Comments on Vibration

Thus the general area on vibration testing is also an area with many problems still awaiting solution. Vigness[80] reported as much in his summary comments following a panel concerning vibration test standardization. He urged standardization wherever possible and study to develop improved test and specifications.

We therefore are obligated to continually develop improved procedures. By encouraging investigations of possible solutions to the many problems we will acquire further knowledge. The knowledge must then be prudently incorporated into the specifications.

PROBLEMS COMMON TO BOTH SHOCK AND VIBRATION

Multidirectional Loading in Shock and Vibration

Several of the topics studied have equal applicability to both the shock and the vibration section and so they are presented in the remainder of the report separately. For example, the ramifications of exciting unidirectionally in three separate mutually perpendicular directions as opposed to the seemingly more realistic simultaneous multidirectional loading are equally applicable to both tests and hence are considered here.

Shock and vibration testing is almost always done separately in three mutually perpendicular directions, as opposed to simultaneous

multidirectional excitation. Notable exceptions are the Navy Medium Weight Shock test in which the equipment is given a set of vertical blows and then a set of obliquely oriented blows, as well as the heavyweight barge shock tests in which the equipment is loaded multidirectionally as the explosive charge hits the barge, much the same as the actual situation. Morrow[4] discusses this problem and logically comments on the effects of unidirectional testing. One is left with the impression that he considers it beyond the state of the practical art and, therefore, somewhat extravagant. Indeed with so many questions unanswered in unidirectional test, it may well seem a little premature to consider multidirectional effects.

Two main reasons motivate us to seek a test that could take into account the multidirectional effects. First, considerable economy would accrue if the three separate tests could be accomplished simultaneously; they would be accomplished in one-third the time as well as without separate refixturing or setup between tests. Second, a more realistic simulation would be obtained. Interaction effects would be considered; that is the possibility of the excitation in one direction causing a response that would interact with a response from another direction to cause failure.

There is a further geometric effect that can be appreciated by the following. Consider a cube in space with one corner designated as the origin of a mutually perpendicular coordinate system; the edges eminating out from that corner represent the three excitation directions. Now consider a round beam cantilevered from the origin of our coordinate system in the cube and oriented along the diagonal, pointing to the opposite corner of the cube. Who is to say and prove that the sum of three excitations along the cube edges is more severe or less severe, than, say excitation in three other mutually perpendicular directions oriented such that two of the directions are perpendicular to the beam axis? In cannot be done. The only real answer lies in the most accurate simulation of the environment, i.e., the unidirectional test would be the most accurate test for equipment used in an unidirectional environment.

Little serious effort has been expended in studying this problem. One particularly amusing pitfall has been documented by Panariti[81] who envisions accomplishing a three-dimensional test by exciting in the diagonal direction of the cube mentioned above. As discussed above, this does not accomplish a three-dimensional test. Consider excitation in the diagonal direction along the beam axis. The diagonal test is a unidirectional test in the diagonal direction. A vector or algebraic combination of three vibrations in mutually perpendicular directions will quickly show that these only resolve to single diagonal vector when the three are related in phase by multiples of 180^{o}. Otherwise a multitude of three dimensional Lissajous' figures result, each of which may be some interesting special test of little more theoretical value than its predecessor. It appears that for any three dimensional test to be truly three dimensional that the phasing between the vectors must vary randomly throughout the test.

Morrow[4] discusses only these multidirectional geometrical effects and significantly notes that depending upon the orientation of the oscillator's response vector, it may be being excited by motion in more than one direction; e.g., an oscillator inclined at 45° to the excitation direction is excited at 0.707 times the applied level for twice the number of cycles, sequentially. He merely poses the problem, and offers as consolation, "Experience suggests that sequential tests. . .lead to satisfactory reliability." One has little chance to question experience; however, we are inclined to believe simultaneous testing is a closer approximation to actual environments although the author has found no reports of three-dimensional environmental measurements to support this feeling.

An effort to design a six degree-of-freedom vibrating exciter has been described[82] with certainly what appears to be a workable scheme, but no studies have been found that present actual experience with multidirectional testing.

Since in the interim between now and the time when we can theoretically predict all the various aspects of dynamic testing, we will be forced time and again to resort to the argument that this or that procedure more accurately simulates the environment; we feel that three dimensional excitation should be studied for possible use.

Blast Loading and Repetitive Shock

Blast loading such as that due to a near miss explosion in the atmosphere or the shock wave eminating from the muzzle of a large cannon, and repetitive shock such as the dynamic field transmitted to the foundation by rapid firing cannon are the two new environments being considered for inclusion in dynamic specifications. Time has not thus far permitted study of these topics. A collection of reports on blast loading has been found in Part IV of Shock and Vibration Bulletin No. 37. The recent study by R. H. Chalmers[83] appears to be the only report treating repetitive shock. This work will be studied for inclusion in the final recommendations of Phase I of this project.

Combined Shock and Vibration Tests

One of the main points cited in the original objectives of this study was to note similarities and redundancies that exist between shock and vibration tests and to see if these cannot be exploited to accomplish economy in testing. The literature survey as far as it has progressed has not turned up a great deal.

MacDuff[84] has studied the use of transient tests to determine natural frequencies, a quantity usually determined by vibration testing. Sneddon[85] shows that Fourier transform theory does rigorously prove that this can be accomplished for linear systems. MacDuff's work is based upon some studies by NACA[86] to determine the natural frequencies of airplanes from their gust response. A distinct contribution of MacDuff's

study is to demonstrate that the methods for linear systems remain reasonably accurate in the presence of nonlinearities. He is presently using this technique to determine the dynamic characteristics of man.

Three papers were found that discuss the subject of using shock to accomplish vibration testing. Villasenor and Butler[87] present an article with a most inviting title: "Use of Shock for Low Frequency Vibration Testing." However, this article merely delves into some complicated shock pulses that will contain appreciable low frequency in their spectra, and does no more. They describe a computer program that enables them to obtain both low and high frequencies in their resulting computed shock spectrum, and also refer to shock spectra with both positive and negative values which will have to be examined further. The shock spectrum is normally taken to be the absolute value of the individual displacements. One must admit, however, that an impact has the potential of exciting a great number of the modes from each point of impact, and as such could indeed generate most of the vibrations of interest, transiently. Noiseux and Watters[88] have examined the resulting quasi-steady-state vibrations that are generated by repetitive impacts similar to that which would be generated by a small riveting hammer. They report that the response so generated appears random in nature and fits some currently specified random vibration spectra.

Bailie[89] also describes a situation where he feels shock to be an adequate substitution for random vibration. He considers the case where the failures are not caused by fatigue but rather by the single highest peak as in brittle type failures. He is referring to a type of non-stationary random vibration which comes about from firing of small rocket motors, staging of launch vehicles, rapid passage through turbulent regions, earthquakes and the like; all of these seem to have been previously characterized by shock. He contends the failures to be due to the single highest peak, even though the loading itself resembles a short burst of random vibrations; thus it is better to test with a shock pulse that has a spectrum that will envelop the single highest peaks rather than random vibrations.

While these ideas do involve loading mechanisms somewhat afield of the actual mechanisms and appear to be opposed to the basic argument of simulating the environment, they do lead us to ponder the question of the extent to which the vibration test can be accomplished by shock or a sequence of low level repetitive shocks.

One distinct possiblity for economy due to test combination is the practice of using electromagnetic shakers to develop shock pulses.[78] This is currently being carried out at Lockheed, Mugu, Boeing and Hughes, to mention a few. This procedure at least permits the shock and vibration tests to be accomplished on a single machine and with a single test fixture. The head of the environmental laboratory at Mugu is very impressed with his accomplishments along these lines and plans to spend developmental money in improving this capacity. Since electromagnetic shakers are limited in maximum displacement (usually 1 inch) only quite low level shocks are currently available with this technique. The idea

of economy of both tests on the same machine is set forth by Painter;[78] he gives a presentation of the general method of producing shock on a shaker. The method is not directly applicable to the general shock test required for shipboard equipment because of its low level. It may well be that future development in improving the displacement characteristics of the electromagnetic shakers or increasing the frequency response of hydraulic shakers may enable the Navy to apply this method to at least lightweight equipment. Due to the huge energy transfer during shock, it is doubted that these developments will be quickly forthcoming, especially since mechanical impact delivers these short bursts of energy so efficiently.

In summary, it is not apparent at this time how optimistic one would dare to be as regards to finding a single test that effectively accomplishes the current shock and vibration tests. It does appear that single machines can be built that can accomplish both tests with the same fixturing.

Hydraulic Shakers

Since the shipboard situation is definitely a low frequency environment, one may well anticipate that electrodynamic shakers may be displacement limited to provide sufficient amplitude at the lower frequencies. The reaction type mechanical machines have control and distortion problems. Therefore it has seemed wise, both in the light of our past experience in the design of hydraulic vibration systems, and with potential need for this type of vibration excitation, to at least accumulate references which discuss the potentialities and limitations of these types of vibration exciters.

MB Electronics is one vibration equipment supplier who has apparently produced hydraulic vibration equipment and hence has cognizance of the field. One of their people, Sipfle[90] has presented an article in which he describes the potentialities of these shakers. Strandrud[91] has described some work done at Boeing in which they have demonstrated a potential for using hydraulic vibration excitation to 1000 cps using hydraulic fluids whose viscosity can be controlled by an electric field thereby affording a new method of control. Ashley [92] presents an analysis of these devices and presents transfer functions useful to predict response as well as an extensive bibliography.

Some concern is definitely indicated here. In conversations with Boeing test personnel,[44] the author was informed of trouble they had encountered with waveform distortion in this type of shaker. Further, one notes in reading Unholtz's discussion of Hydraulic Vibration Machines in Harris and Crede,[1] a reference to waveform distortion inherent in these machines. It appears to be a problem that is not too publicly discussed, and hence would certainly warrant investigation if hydraulic actuators were incorporated into any design of new equipment. A very good start would be to contact MB Electronics, proponents of these machines, and inquire about their problems and capabilities in this area.

NCEL is currently embarking on the final design stages of a combination dynamic test machine that will ultimately be capable of performing two-dimensional shock and vibration simultaneously. In FY-71 we plan to fabricate and test the hydraulic shaking mechanism and thus determine the state-of-the-art in developing low frequency vibration via hydraulic actuators. It may well turn out that the work of the two laboratories in combination may ultimately develop at least a machine capable of considerable economy in testing.

Failure Definition

An important matter to consider is that which constitutes failure of the dynamic test. One must specify if the equipment is to operate during or after the test, especially with regard to shock tests. Comprehensive consideration of the total function of the equipment must be considered with regard to failure definition. Exact tests to determine performance during and after the test must be specified in the test plan. Limits on functional degradation and the fine line constituting failure should be well defined. A formal failure definition and means of perceiving failure must be agreed upon prior to test. Emphasis must be placed upon this point to assure that only highly qualified personnel, with a complete grasp of the total function of the equipment to be tested, be given this responsibility of determining the failure definition.

CONCLUSIONS

In summary, the results of a general literature survey, together with comments on interviews with test personnel, have been organized to indicate the breadth and depth of the field of dynamic test. A great deal more literature is available, and even more has arrived since this report was written. This first report is offered to show the direction and progress of the study so that future efforts may have the benefit of NELC's opinions.

The following points seem to develop as a result of the study thus far:

(a) Increased research in shock definition and measurement is required.

(b) Monitoring and hence measurement should be incorporated into the tests as the state of the art progresses and permits.

(c) Impedance effects and fixtures must be researched and the progress incorporated into the tests.

(d) Shock damage theory should be developed to permit economy in both measurement and testing so that only the essential potentially damaging portion of the shock need be applied as the test load.

(e) Much more reliable vibration tests are available than the search and dwell procedure currently specified in MIL-STD-167.

(f) Navy vibration environments should be studied to assess the degree of randomness and hence the extent to which random vibration testing is needed.

(g) Research in vibration equivalence should be continued to determine the ramifications of interchanging swept sine, random, and actual field vibrations.

(h) Multidirectional effects should be experimentally studied to evaluate uniaxial testing sufficiency.

(i) Combined shock and vibration testing at a significant level is currently only available through a combination of state of the art techniques. NCEL has designed such a machine but to date none have been built.

(j) Hydraulic shakers seem to hold great promise for accurate low frequency vibration tests. They should be evaluated for possible Naval utilization.

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13. ABSTRACT

The report presents a review of current shock and vibration testing technology, based upon literature study, conferences with testing personnel and shock and vibration experts, and the personal experiences of the author. The technological frontiers of the many aspects of shock and vibration testing are outlined. Those areas of research which would most significantly improve reliability or economy are indicated. The report also indicates those practices and theories which are not based upon fact and necessarily involve uncertainties.

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